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ELASTIC-PLASTIC ANALYSIS OF ANNULAR PLATE PROBLEMS USING NASTRAN

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US ARMY ARMAMENT RESEARCH AND DEVELOPMENT COMMAND
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The plate elements of the NASTRAN code are used to analyze two annular plate problems loaded beyond the elastic limit. The first problem is an elastic-plastic annular plate loaded externally by two concentrated forces. The second problem is stressed radially by uniform internal pressure for which an exact analytical solution is available. A comparison of the two approaches together with an assessment of the NASTRAN code is given.

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INTRODUCTION

The piecewise linear analysis option of the NASTRAN code can be used to analyze quite complicated elastic-plastic plane-stress problems.[1,2] The reliability of this code has been well demonstrated in the linear range but not so in the nonlinear range of loadings. One major reason is because exact analytical solutions for elastic-plastic problems are usually not available for comparison with approximate NASTRAN solutions.

In this report, the plate elements of the NASTRAN code are used to solve two annular plate problems loaded beyond the elastic limit. The first problem considered is an elastic-plastic annular plate loaded externally by two concentrated forces. There is no analytical solution for this two-dimensional plane-stress problem and the NASTRAN code is used to obtain numerical results. The second problem considered is an elastic-plastic annular plate radially stressed by uniform internal pressure. This problem is chosen because an exact analytical solution is available for comparison. For ideally plastic materials, the stress solution for this statically determinate problem was first obtained by Mises[3] and the corresponding two strain solutions were obtained by the present author on the basis of both J_2 deformation and flow theories.[4] For elastic-plastic strain-hardening materials, an exact complete

[1]NASTRAN Demonstration Problem Manual, NASA SP-224 (01), 1972.
[2]Chen, P. C. T., "On Elastic-Plastic Analysis of an Overloaded Breech Ring Using NASTRAN," Proceedings of the Tenth NASTRAN Users' Colloquium, May 1982, pp. 100-109.
[3]Mises, R. V., "Three Remarks on the Theory of Ideal Plastic Body," Reissner Anniversary Volume, 1949, pp. 415-429.
[4]Chen, P. C. T., "A Comparison of Flow and Deformation Theories in a Radially Stressed Annular Plate," Journal of Applied Mechanics, Vol. 40, No. 1, Trans. ASME, Vol. 95, 1973, pp. 283-287.

solution was recently reported in Reference 5. Analytical expressions were derived on the basis of J_2 deformation theory, the Hill's yield criterion, and a modified Ramberg-Osgood law. The validity of the above solution has been established by satisfying the Budiansky's criterion.

In the following paragraphs, the theory of elastic-plastic plate elements as used in NASTRAN is briefly reviewed. In its present form, NASTRAN cannot be used for problems involving ideally plastic materials. It is shown that this limitation can be easily removed by making minor changes. For an elastic-plastic strain-hardening material, the NASTRAN solution is reported here and compared with the exact solution. The results are presented graphically and an assessment of the NASTRAN code is made.

PLASTIC PLATE ELEMENTS

The theoretical basis of two-dimensional plastic deformation as used in NASTRAN is that developed by Swedlow[6]. In the development, a unique relationship between the octahedral stress, τ_0, and the plastic octahedral strain, $\varepsilon_0{}^P$, is assumed to exist and the use of ideally plastic materials is excluded. The total strain components (ε_x, ε_y, ε_z, and γ_{xy}) are composed of the elastic, recoverable deformations and the plastic portions ($\varepsilon_x{}^P$, $\varepsilon_y{}^P$, $\varepsilon_z{}^P$, and $\gamma_{xy}{}^P$). The rates of plastic flow, ($\dot{\varepsilon}_x{}^P$, etc.), are independent of a time scale and are simply used for convenience instead of incremental values. The

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[5]Chen, P. C. T., "Elastic-Plastic Analysis of a Radially Stressed Annular Plate," Journal of Applied Mechanics, Vol. 44, 1977, pp. 167-169.
[6]Swedlow, J. J., "The Thickness Effect and Plastic Flow in Cracked Plates," ARL 65-216, Aerospace Research Laboratories, Office of Aerospace Research, October 1965.

definitions of the octahedral stress and the octahedral plastic strain rate

for isotropic materials are:

$$\tau_0 = \sqrt{(s_{11}^2 + 2s_{12}^2 + s_{22}^2 + s_{33}^2)/3} \tag{1}$$

$$\dot{\varepsilon}_0^P = \sqrt{[(\dot{\varepsilon}_{11}^P)^2 + 2(\dot{\varepsilon}_{12}^P)^2 + (\dot{\varepsilon}_{22}^P)^2 + (\dot{\varepsilon}_{33}^P)^2]/3} \tag{2}$$

where

$$s_{11} = \frac{1}{3}(2\sigma_x - \sigma_y) \quad , \quad \varepsilon_{11}^P = \varepsilon_x^P$$

$$s_{22} = \frac{1}{3}(2\sigma_y - \sigma_x) \quad , \quad \varepsilon_{22}^P = \varepsilon_y^P$$

$$s_{33} = -\frac{1}{3}(\sigma_x + \sigma_y) \quad , \quad \varepsilon_{33}^P = \varepsilon_z^P$$

$$s_{12} = T_{xy} \quad , \quad \varepsilon_{12}^P = \frac{1}{2}\gamma_{xy}^P \tag{3}$$

The s_{ij} is called the deviator of the stress tensor; σ_x, σ_y, and T_{xy} are the

cartesian stresses. The isotropic material is assumed to obey the Mises yield

criterion and the Prandtl-Reuss flow rule. The matrix relationship for the

plastic flow is:

$$
\begin{Bmatrix} \dot{\varepsilon}_x^P \\ \dot{\varepsilon}_y^P \\ \dot{\gamma}_{xy}^P \end{Bmatrix} = \frac{1}{6\tau_0^2 M_T(\tau_0)}
\begin{bmatrix} s_{11}^2 & s_{11}s_{22} & 2s_{11}s_{12} \\ s_{11}s_{22} & s_{22}^2 & 2s_{22}s_{12} \\ 2s_{11}s_{12} & 2s_{22}s_{12} & 4s_{12}^2 \end{bmatrix}
\begin{Bmatrix} \dot{\sigma}_x \\ \dot{\sigma}_y \\ \dot{T}_{xy} \end{Bmatrix} \tag{4}
$$

where

$$2M_T(\tau_0) = \dot{\tau}_0/\dot{\varepsilon}_0^P \tag{5}$$

The plastic modulus, $M_T(\tau_0)$, can be related to the slope E_T, of the effective

stress-strain curve by:

$$\frac{1}{3M_T(\tau_0)} = \frac{1}{E_T} - \frac{1}{E} \tag{6}$$

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The total strain increments, obtained by adding the plastic and linear elastic parts, are:

$$\{\Delta\epsilon\} = ([D^P] + [G]^{-1})\{\Delta\sigma\} = [G_p]^{-1}\{\Delta\sigma\} \tag{7}$$

where $[G]$ is the normal elastic material matrix and $[G_p]$ is the equivalent plastic material matrix. The matrices $[D^P]$ and $[G_p]^{-1}$ exist for finite values of M_T or (E_T) and $[G_p]$ can be obtained numerically. Because this procedure is chosen in developing NASTRAN program, only strain hardening materials can be considered for applications. However, it should be noted that even the matrix $[Gp]^{-1}$ does not exist when M_T or E_T is equal to zero, the matrix $[G_p]$ may still exist. In fact, the closed form of $[G_p]$ has been given in Reference 7. We can express this as:

$$[G_p] = \frac{E}{Q} \begin{pmatrix} s_{22}^2 + 2A & & & \text{SYM.} \\ -s_{11}s_{22} + 2\nu A & , & s_{11}^2 + 2A & & \\ -\dfrac{s_{11} + \nu s_{22}}{1+\nu}s_{12} & , & -\dfrac{s_{22} + \nu s_{11}}{1+\nu}s_{12} & , & \dfrac{B}{2(1+\nu)} + \dfrac{(1-\nu)E_T}{E - E_T}\tau_0^2 \end{pmatrix} \tag{8}$$

where

$$A = \frac{E_T}{E - E_T}\tau_0^2 + \frac{s_{12}^2}{1+\nu} \quad , \quad B = s_{11}^2 + 2\nu s_{11}s_{22} + s_{22}^2 \quad , \quad Q = 2(1-\nu^2)A + B \tag{9}$$

If we want to remove the limitation that the use of ideally plastic materials is excluded, we have to make minor changes in subroutines PSTRM and PKTRM of the NASTRAN program.

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[7]Yamada, Y., Yoshimura, N., and Sakuri, T., "Plastic Stress-Strain Matrix and Its Application For the Solution of Elastic-Plastic Problems by the Finite Element Method," Int. J. Mech. Sci., Vol. 10, 1968, pp. 345-354.

TWO-DIMENSIONAL PROGRAM

Consider a two-dimensional annular plate loaded externally by two
concentrated forces. Figure 1 shows a finite element representation for one
quarter of the annular plate. The other part is not needed because of
symmetry. There are 198 grids and 170 quadrilateral elements in this model.
The grids (1 through 11) along the x-axis are constrained in y-direction while
those grids (188 through 198) along the y-axis are constrained in the
x-direction. The concentrated force F is applied at the top of the y-axis
(grid 198). The thickness of the plate is 0.1 inch. All membrane elements
are stress dependent materials. The effective stress-strain curve is defined
by:

$$\sigma = E\epsilon \quad \text{for} \quad \sigma < \sigma_0$$

$$(\sigma/\sigma_0)^n = (E/\sigma_0)\epsilon \quad \text{for} \quad \sigma > \sigma_0 \tag{10}$$

where

$$\sigma = (3/\sqrt{2})\tau_0 \quad , \quad \epsilon = \sigma/E + \sqrt{2}\ \epsilon_p \tag{11}$$

n is the strain hardening parameter and the initial yield surface is defined
by the ellipse $\sigma = \sigma_0$. The input parameters for the problem are:

$E = 10.5 \times 10^6$ psi , $\nu = 0.3$ (Poisson's ratio) , $\sigma_0 = 5.5 \times 10^4$ psi , $n = 9$

There is no analytical solution for the two-dimensional plane-stress problem
and the NASTRAN code is used to obtain numerical results. First the stress
solution in the elastic loading range is obtained and the elastic limit is
determined. The corresponding F* at initial yielding is 753.34 pounds. Then
the solution beyond the elastic limit is obtained in 13 steps with the applied
force given by $F_n = 753.36\ (0.95 + 0.05n)$ pounds for $n = 1,2,\ldots13$. The
vertical displacement at the point of application (V_D) and the horizontal
displacement (U_B) at point B (grid 11) are shown in Figure 2 as functions of

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the applied force F. The major principal stresses in elements near the x and y-axes are shown in Figure 3 for F = 600 and 1205.38 pounds, respectively. The results indicate that the maximum tensile stress occurs at point C.

ONE-DIMENSIONAL PROBLEM

Consider a one-dimensional annular plate stressed radially by uniform internal pressure. The plate geometry and material properties are the same as the two-dimensional one. Utilizing the condition of axial symmetry, we need only a sector of the annular plate for the finite element model. There are only 10 elements with 22 grids. All grid points are constrained in the tangential direction. The applied load is the internal pressure p. If p = 1000 psi, the equivalent nodal force at each of the two interior grids is Q = 4.62329 pounds in the radial direction. The true pressure corresponding to initial yielding for this problem is p_o = 23,571 psi. The NASTRAN results will depend on the users' choice of element sizes and load increments. In the elastic range, the NASTRAN results based on two finite-element models were compared with the exact solution. For a ten-element model, the maximum error is 0.36 percent in displacements and 0.95 percent in stresses. For a twenty-element model, the maximum errors are reduced to 0.09 percent and 0.24 percent, respectively. Since we are satisfied with one percent error, the ten-element model is chosen for incremental analysis beyond the elastic limit.

In the plastic range, the user has to choose the load increments properly in order to obtain good results at reasonable cost. The values of the load factors can be normalized so its unit value corresponds to the limit of elastic solution, i.e., p_o = 23,571 psi. The load increments can be uniform or

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nonuniform. It seems that the size of load increments depends on the material properties and sizes of elements. In order to determine the influence of load factors on the displacements and stresses in the plastic range, four sets of load factors are chosen. The load increments for three of them are uniform with $\Delta p/p_o = 0.20$, 0.10, and 0.05, respectively. The influence of load factor, p/p_o, on the inside radial displacement, u_1, is shown in Figure 4. The effect of load factors on the major principal stresses in elements is shown in Figure 5. We also show in these two figures the corresponding analytical results. On the basis of these comparisons, we can make the following conclusions. In the earlier stages of plastic deformation, larger load increments can be used to give very good results. As plastic deformation becomes bigger, smaller load increments should be used in order to get a reasonably good answer. However, for very large plastic deformation, it seems that we cannot improve the NASTRAN results much better by choosing even smaller increments. This is because there are other built-in errors in the NASTRAN program, e.g., the linear displacement function is assumed.

If uniform load increments are used, a direct comparison of the analytical and NASTRAN results is not available. The solid curves shown in Figures 4 and 5 were obtained indirectly since the analytical results of the displacements, stresses, and pressure were represented as functions of elastic-plastic boundary.[5] We have obtained the analytical results when the elastic-plastic boundary is located at a radial distance of 0.05, 0.15, 0.25, 0.35, 0.45, 0.55,

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[5]Chen, P. C. T., "Elastic-Plastic Analysis of a Radially Stressed Annular Plate," Journal of Applied Mechanics, Vol. 44, 1977, pp. 167-169.

0.65, 0.75, 0.85, and 0.95 inch from the inside surface. The corresponding values of the pressure factor (p/p_0) are 1.095, 1.266, 1.414, 1.540, 1.646, 1.734, 1.805, 1.860, 1.901, and 1.929, respectively. This set of values is used as load factors in the input deck of the NASTRAN program. Some of the NASTRAN results together with the corresponding analytical results are shown in Figures 6 and 7. A direct comparison of the two approaches in the plastic range can be seen. The effect of load factors on the distribution of radial displacements is shown in Figure 6. The distributions of major principal stresses for three load factors are shown in Figure 7. As can be seen in Figures 6 and 7, a direct comparison of two approaches will support the following conclusion. Even if the results in the elastic range are in excellent agreement, the differences in the plastic range can be quite big for large values of load factors. This suggests more research efforts should be given to large plastic deformation.

CONCLUSIONS

Two elastic-plastic annular plate problems have been analyzed by using NASTRAN plate elements. One problem is loaded externally by two concentrated forces and the other by uniform internal pressure. The NASTRAN results for the second problem have been compared with an exact analytical solution. It seems that the NASTRAN code in its present form is still a valuable tool because it can be used to solve quite complicated plane stress problems provided the plastic deformation involved is not too large. The limitation that the use of ideally plastic materials is excluded can be easily removed by making minor changes.

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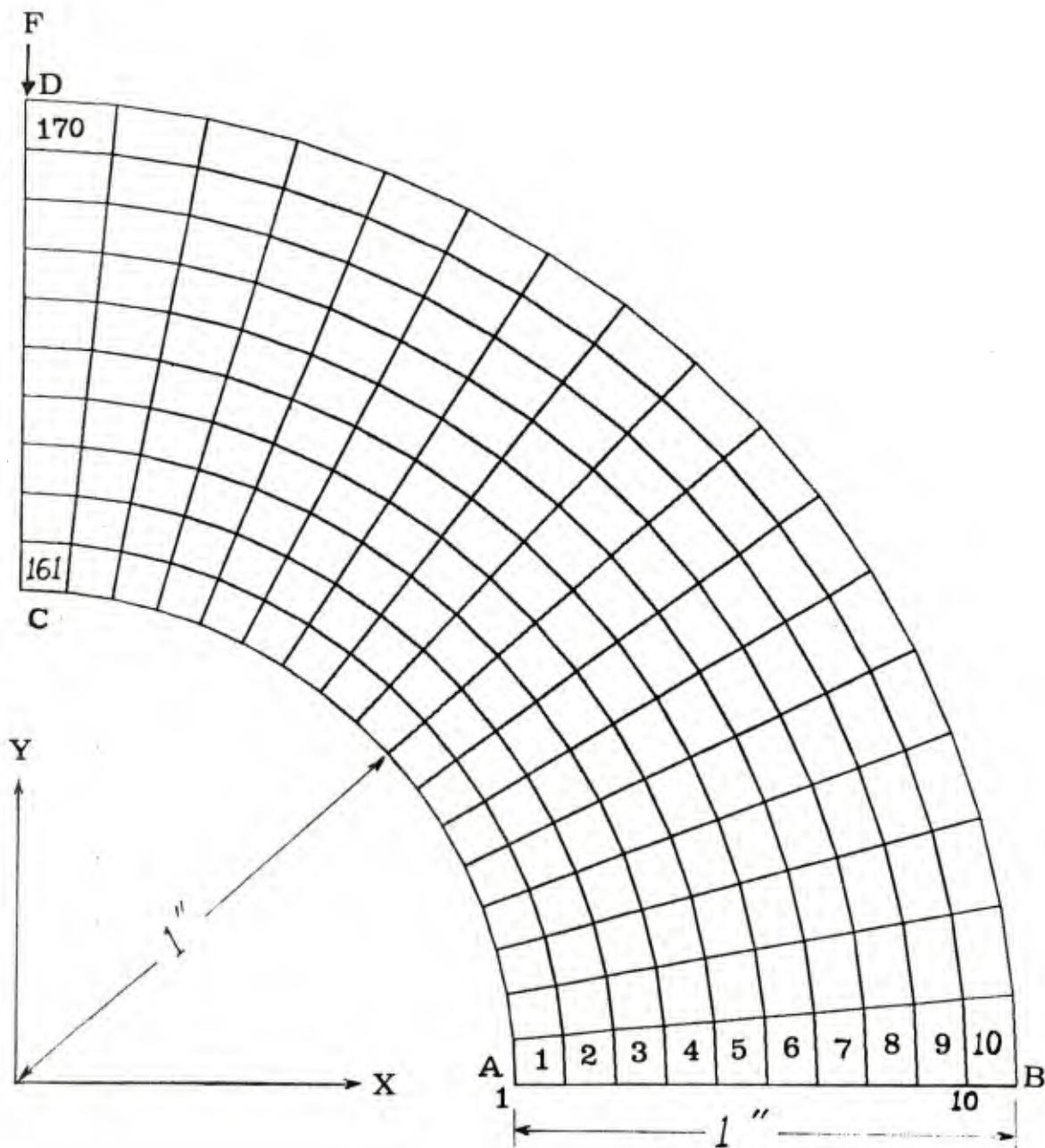
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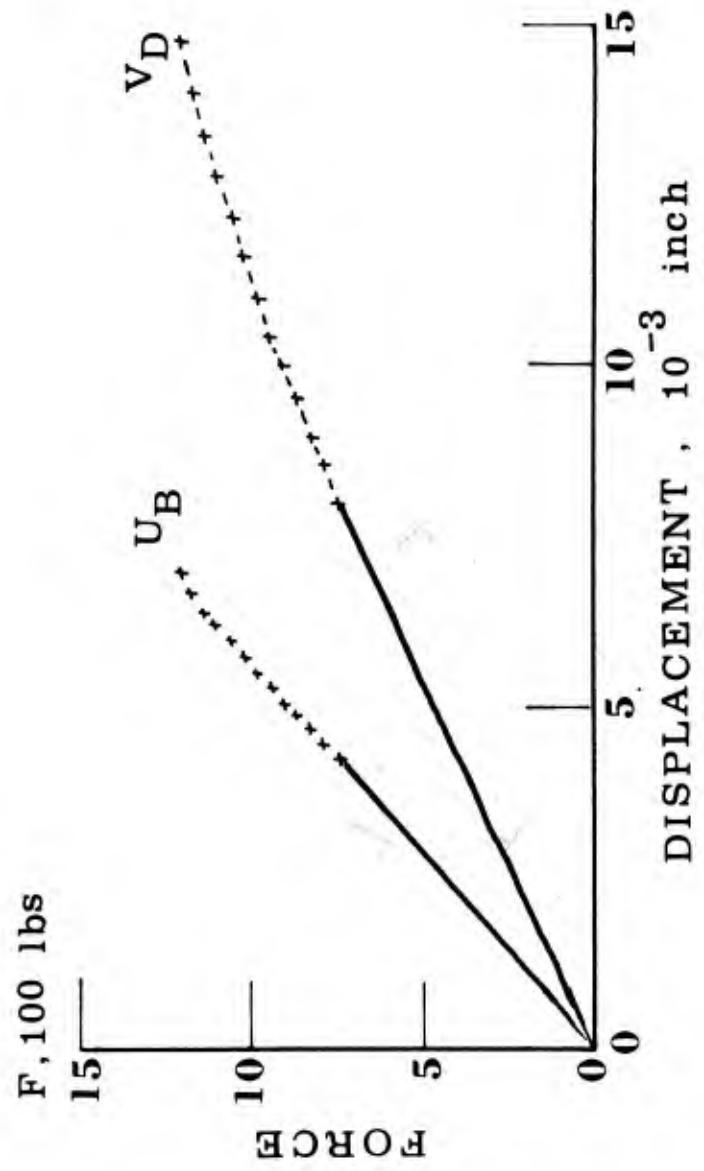
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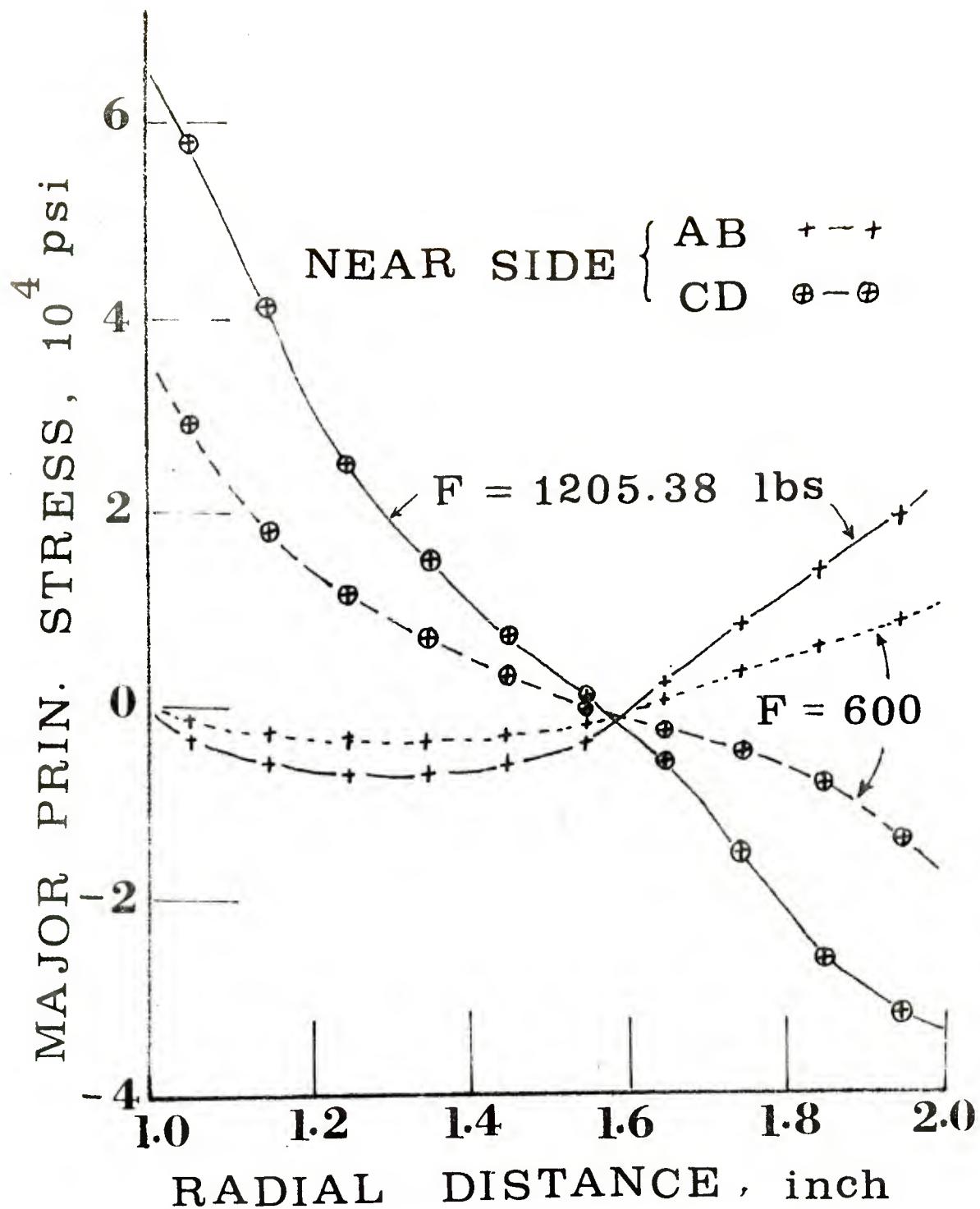
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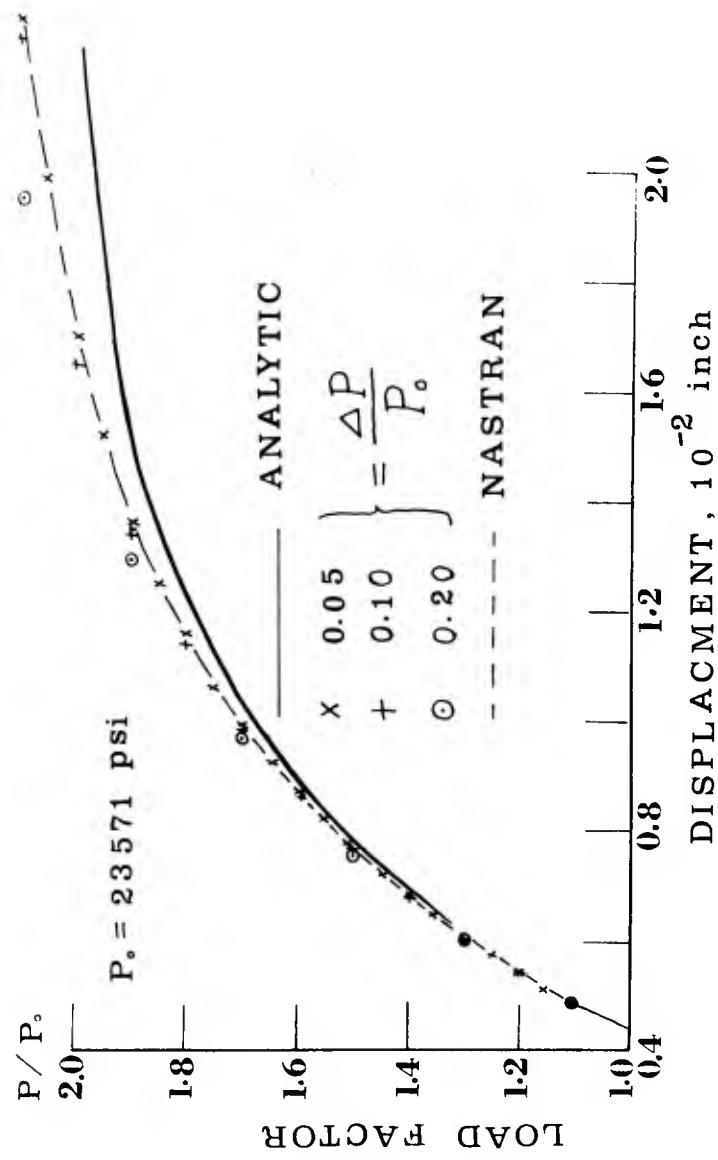
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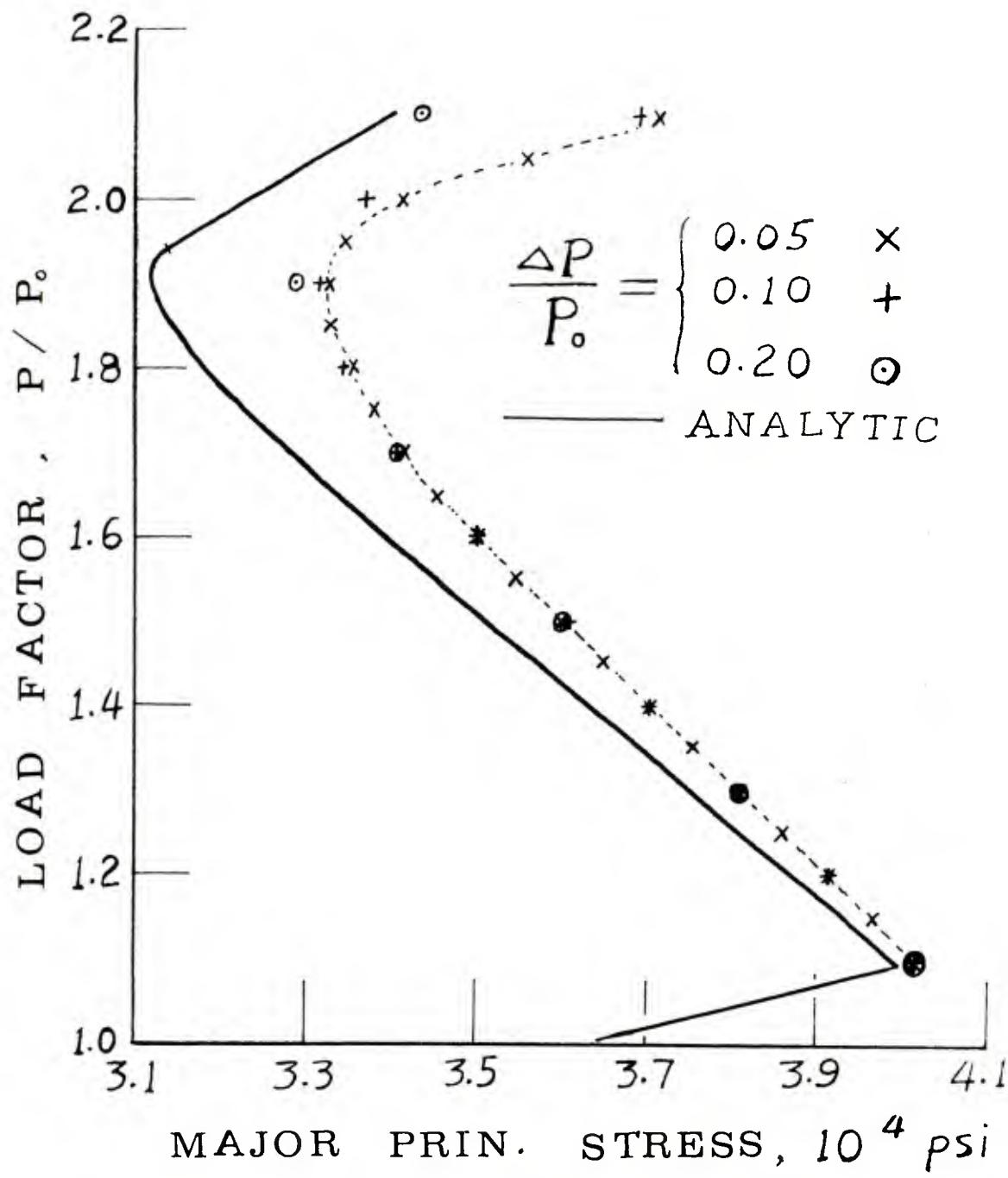
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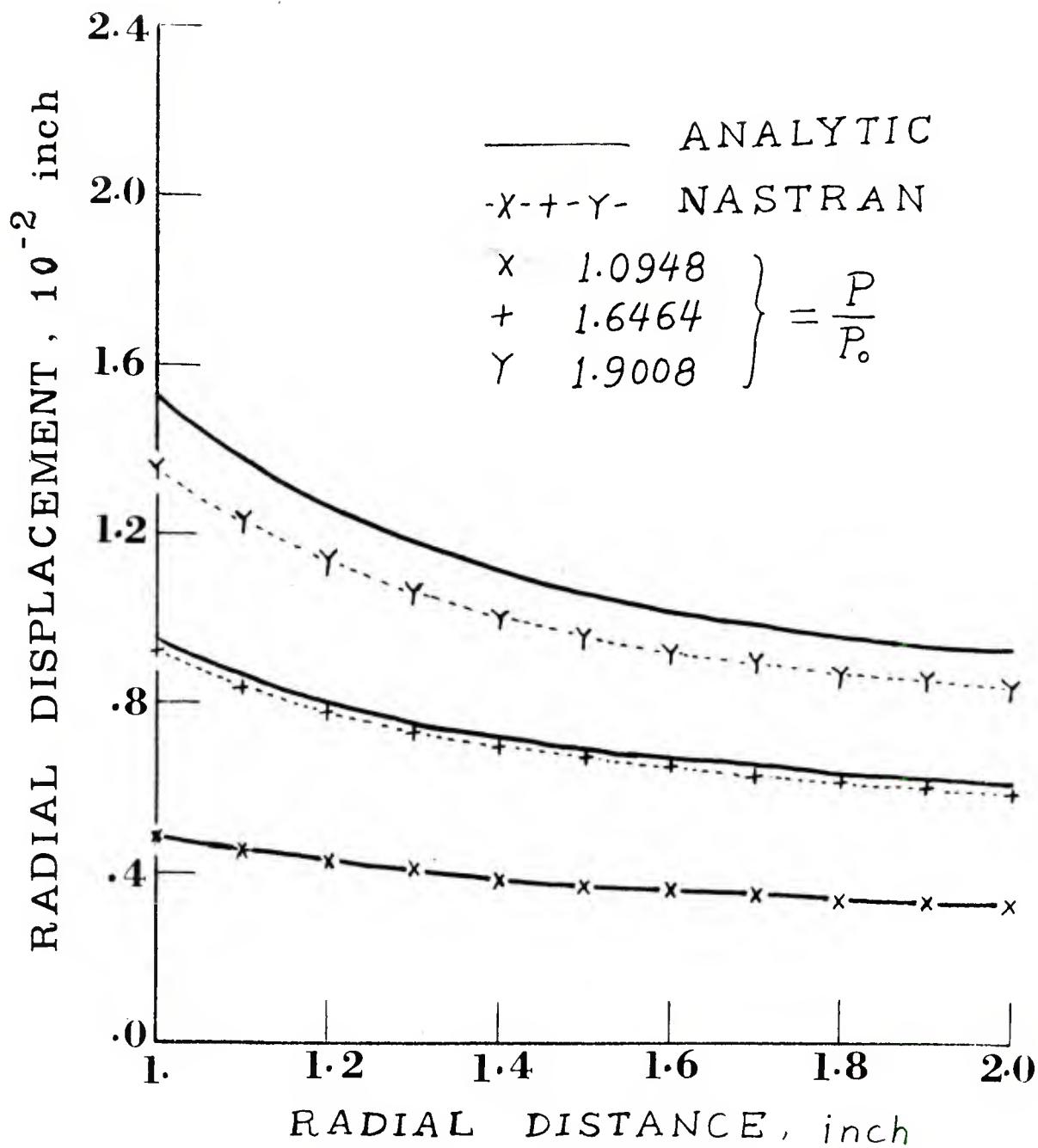
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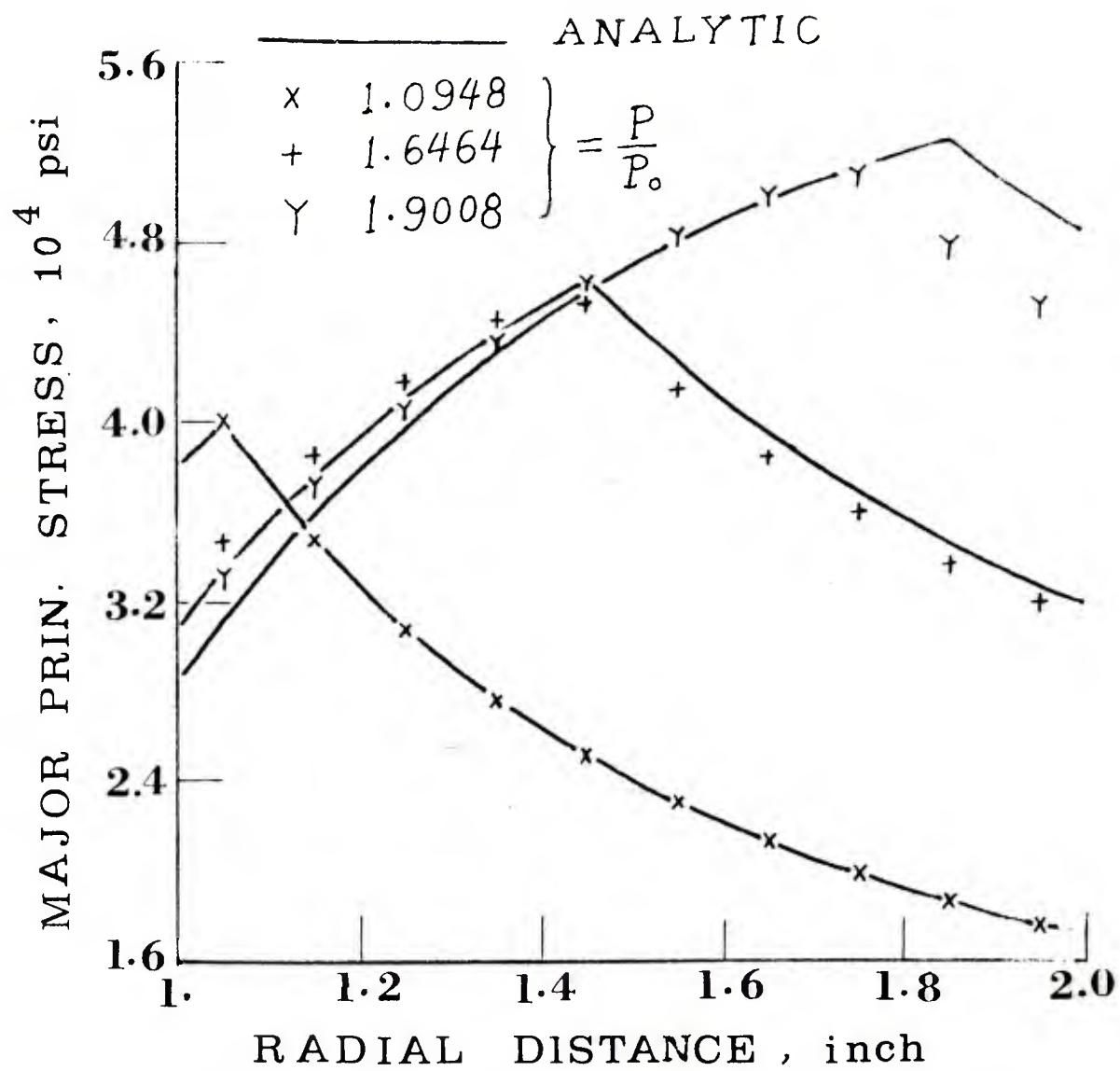
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